GEORGIA INSTITUTE OF TECHNOLOGY
OFFICE OF CONTRACT ADMINISTRATION
SPONSORED PROJECT INITIATION

Date: January 15, 1980

Project Title: "Parametric Study of Heat Pipe Startup from the Supercritical State"

Project No: E-25-626

Project Director: Dr. G. T. Colwell

Sponsor: National Science Foundation

Agreement Period: From 11/15/79 Until 4/30/81

Type Agreement: Grant No. CME-7908414

Amount: $30,662 (E-25-626)
890 (E-25-348 cost-sharing)
$31,552 Total

Reports Required: Annual Progress Report (if renewal is requested); Final Report

Sponsor Contact Person (s):

Technical Matters	Contractual Matters (thru OCA)
Program Officer Win Aung (202) 632-5787	Grants Official Terry J. Pacovsky (202) 632-5892

Defense Priority Rating: N/A

Assigned to: Mechanical Engineering (School/~~Laboratory~~)

COPIES TO:

- Project Director
- Division Chief (EES)
- School/Laboratory Director
- Dean/Director—EES
- Accounting Office
- Procurement Office
- Security Coordinator (OCA)
- Reports Coordinator (OCA)
- Research Property Coordinator (OCA)
- Library, Technical Reports Section
- EES Information Office
- EES Reports & Procedures
- Project File (OCA)
- Project Code (GTRI)
- Other

GEORGIA INSTITUTE OF TECHNOLOGY OFFICE OF CONTRACT ADMINISTRATION

SPONSORED PROJECT TERMINATION SHEET

Date 7/13/82

Project Title: Parametric Study of Heat Pipe Startup from the Supercritical State

Project No: E-25-626

Project Director: Dr. G.T. Colwell

Sponsor: National Science Foundation

Effective Termination Date: 4/30/81

Clearance of Accounting Charges: 4/30/81

Grant/Contract Closeout Actions Remaining:

- [ ] Final Invoice and Closing Documents
- [x] Final Fiscal Report (FCTR)
- [ ] Final Report of Inventions
- [ ] Govt. Property Inventory & Related Certificate
- [ ] Classified Material Certificate
- [ ] Other ____________

Assigned to: Mechanical Engineering (School/~~Laboratory~~)

COPIES TO:

Administrative Coordinator
Research Property Management
Accounting
Procurement/EES Supply Services
Research Security Services
~~Reports Coordinator~~ (OCA)
Legal Services (OCA)
Library
EES Public Relations (2)
Computer Input
Project File
Other ____________

FORM OCA 10:781

E-25-626

FINAL REPORT

Period Covering November 15, 1979 to April 30, 1981

PARAMETRIC STUDY OF HEAT PIPE STARTUP FROM THE SUPERCRITICAL STATE

Prepared for

THE NATIONAL SCIENCE FOUNDATION
Heat Transfer Program
Grant CME-7908414

July 15, 1981

GEORGIA INSTITUTE OF TECHNOLOGY
A UNIT OF THE UNIVERSITY SYSTEM OF GEORGIA
SCHOOL OF MECHANICAL ENGINEERING
ATLANTA, GEORGIA 30332

1981

PARAMETRIC STUDY OF HEAT PIPE STARTUP FROM THE SUPERCRITICAL STATE

FINAL REPORT

to

The National Science Foundation
Heat Transfer Program

Grant CME-7908414
November 15, 1979 to April 30, 1981

BY

The School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, Georgia 30332

July 15, 1981

TABLE OF CONTENTS

FINAL REPORT

National Science Foundation
Heat Transfer Program

Grant CME-7908414

Endorsements:

Gene T. Colwell
Principal Investigator

S. Peter Kezios, Director
School of Mechanical Engineering

William M. Sangster
Dean of Engineering

Dwight L. Allen
Deputy Director
Contract Administration

NATIONAL SCIENCE FOUNDATION
Washington, D.C. 20550

FINAL PROJECT REPORT
NSF FORM 98A

PLEASE READ INSTRUCTIONS ON REVERSE BEFORE COMPLETING

PART I–PROJECT IDENTIFICATION INFORMATION

1. Institution and Address	2. NSF Program	3. NSF Award Number
Georgia Institute of Technology School of Mechanical Engineering Atlanta, Georgia 30332	Heat Transfer	CME-7908414
	4. Award Period From 11/15/79 To 4/30/81	5. Cumulative Award Amount $31,552

6. Project Title

Parametric Study of Heat Pipe Startup From The Supercritical State

PART II–SUMMARY OF COMPLETED PROJECT *(FOR PUBLIC USE)*

(See Attached Sheets)

PART III–TECHNICAL INFORMATION *(FOR PROGRAM MANAGEMENT USES)*

1. ITEM *(Check appropriate blocks)*	NONE	ATTACHED	PREVIOUSLY FURNISHED	TO BE FURNISHED SEPARATELY TO PROGRAM Check (√)	TO BE FURNISHED SEPARATELY TO PROGRAM Approx. Date
a. Abstracts of Theses		X			
b. Publication Citations		X			
c. Data on Scientific Collaborators					
d. Information on Inventions	X				
e. Technical Description of Project and Results		X			
f. Other *(specify)* Significant Research Accomplishments		X			
g. Personnel Supported		X			

2. Principal Investigator/Project Director Name *(Typed)*	3. Principal Investigator/Project Director Signature	4. Date
Gene T. Colwell		

NSF Form 98A (5-78) Supersedes All Previous Editions

Form Approved OMB No. 99R0013

II. Summary of Completed Project

This project consisted of one year continuation of an original two year project aimed at studying experimentally and theoretically transient operation of heat pipes. Work performed in the original project included building a well instruemented heat pipe test section, making measurements of transient operation under a variety of operating conditions, and developing a mathematical model and related computer program which could be used to predict transient capillary structure performance. The goal of the continuation project was to gather and reduce experimental data and data generated with aid of the computer program and to develop simplified correlations based on this data.

Results of the present study indicate that transient heat pipe performance may be broadly classified as (1) normal heating and cooling, (2) heating where a portion of the evaporator capillary structure dries but where steady state operation eventually occurs with modest internal pressures and temperatures, (3) heating where the entire evaporator section eventually dries and excessive internal pressures and temperatures occur, and (4) operation where rewetting of the capillary structure occurs. With normal heating and cooling, the capillary structure remains fully wetted, the heat pipe is nearly internally isothermal at any instant of time, and internal operation may be modeled as a single lumped thermal mass. When partial drying occurs in the evaporator region, large internal temperature gradients may occur and the model must account for liquid dynamics in the capillary structure. No modeling is required for classification (3) since expressions for predictin capillary limits are well known. During restarting after a significant portion of the capillary structure has dried, the transient operation is controlled by liquid flow dynamics which depend upon surface tension forces, resistance to flow, and interfacial evaporation.

Results obtained during both grants are being presented in the open literature.

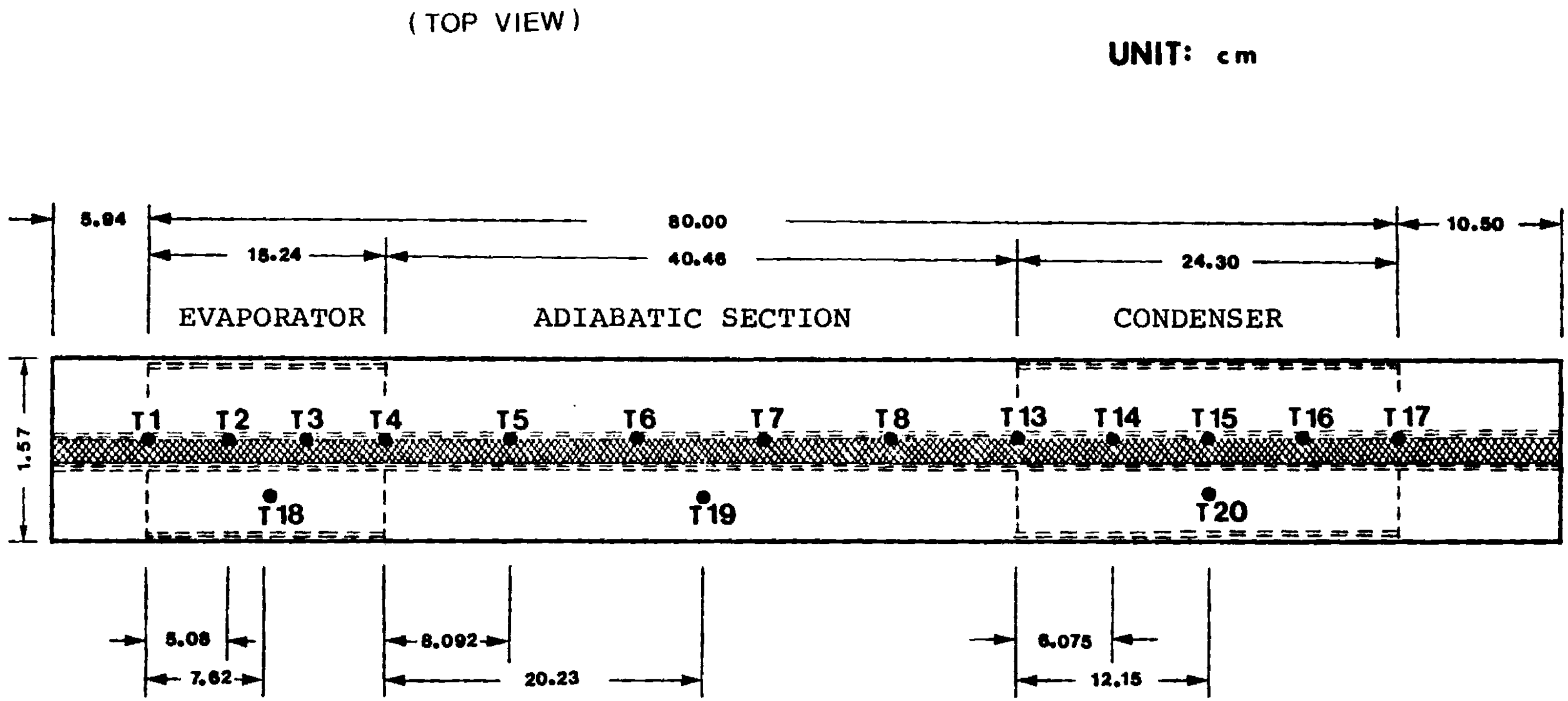
III. e. Technical Description of Project and Results

The purposes of this one year project were to reduce experimental and theoretical data obtained during a preceeding two year project and to develop simplified correlations which may be used to estimate transient heat pipe performance under a variety of operating conditions.

Experimental Data

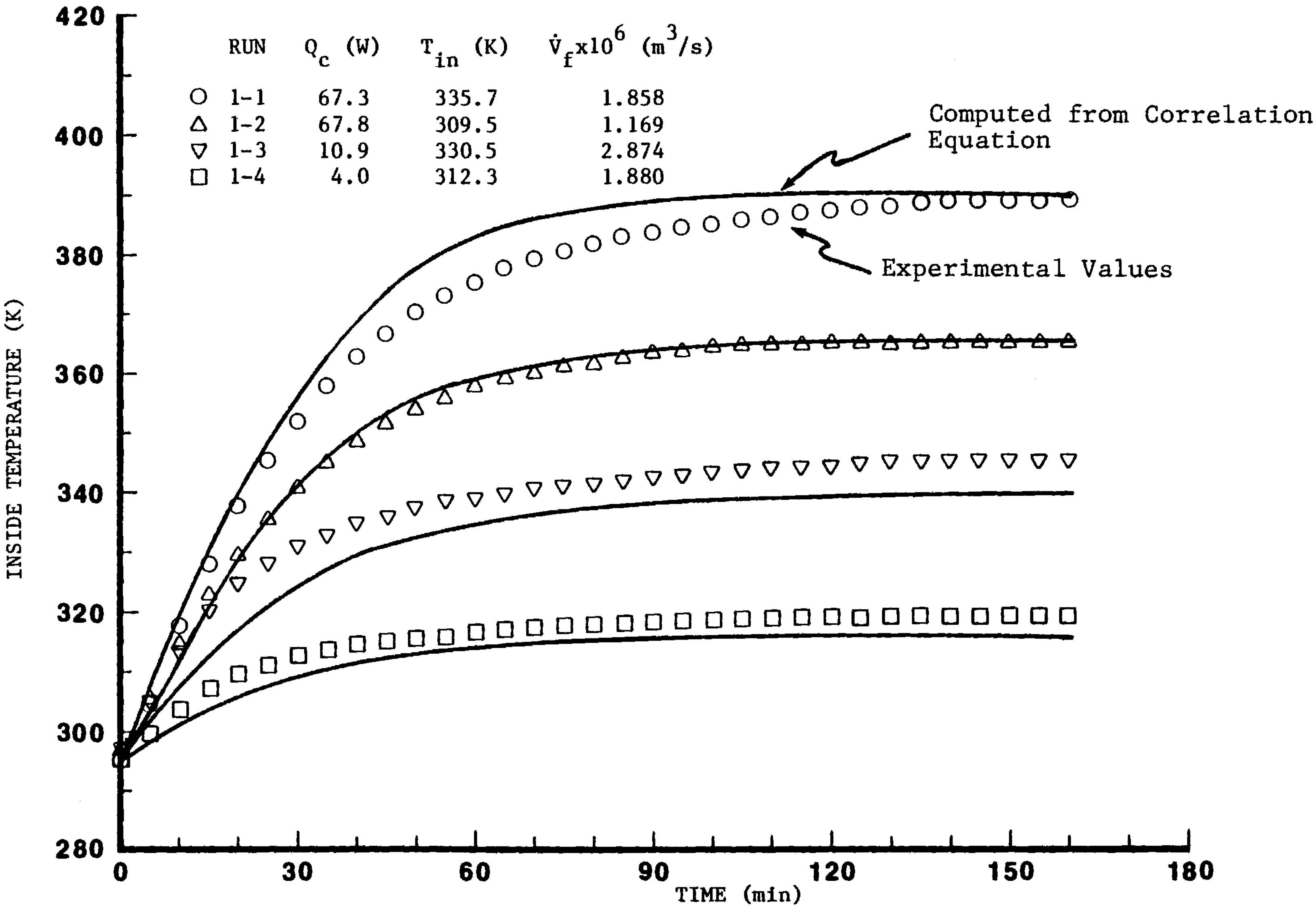
A well instrumented stainless steel heat pipe 80 cm in length and 1.9 cm in outside diameter was operated with freon 11. The capillary structure consisted of two circumferential layers of 100 mesh stainless steel screen along the inside wall of the pipe in evaporator and condenser sections and a central slab consisting of four layers of 40 mesh stainless steel screen covered on both sides by two layers (four layers total) of 100 mesh screen. Energy was supplied by electrical heating and the device was cooled by circulating a silicone fluid through a cooling jacket. The internal location of thermocouples is shown in Figure 1.

The experimental data gathered may be sorted into four categories. The first category includes all "normal" heat pipe operation where the capillary structure remains fully wetted during transients and where the heat pipe is internally very nearly isothermal at an instant of time. Figure 2 shows some data of this type. The symbols indicate measured internal temperatures (which were very closely uniform throughout the interior of the heat pipe at an instant



Inside Thermocouple Locations

Figure 1



Transient Temperature Responses under Various Operating Conditions (Runs 1-1 through 1-4)

Figure 2

of time) for a variety of conditions. The curves represent results of computer computations which will be described later. Q_c represents heat flow from the condenser, T_{IN} represents inlet coolant temperature and $\dot{V}_f$ represents coolant flow rate. For all four cases shown the heat pipe was initially at room temperature with no heat input and no coolant flow. At time zero, heat was suddenly applied and coolant was suddenly introduced to the cooling jacket. All operating parameters smoothly approached their steady operating values during the transient period. Similar performance was obtained for transient cooling operation.

Category two includes those cases where the capillary structure in the evaporator partially dries but where steady operation occurs without excessive pressures or temperatures even though rather large internal temperature differences may develop. Figure 3 shows such a case where axial temperature distributions are given as a function of time for transient heating. Notice that after about 100 minutes drying occurs in the evaporator. However, under these conditions the drying is contained and steady operation is reached.

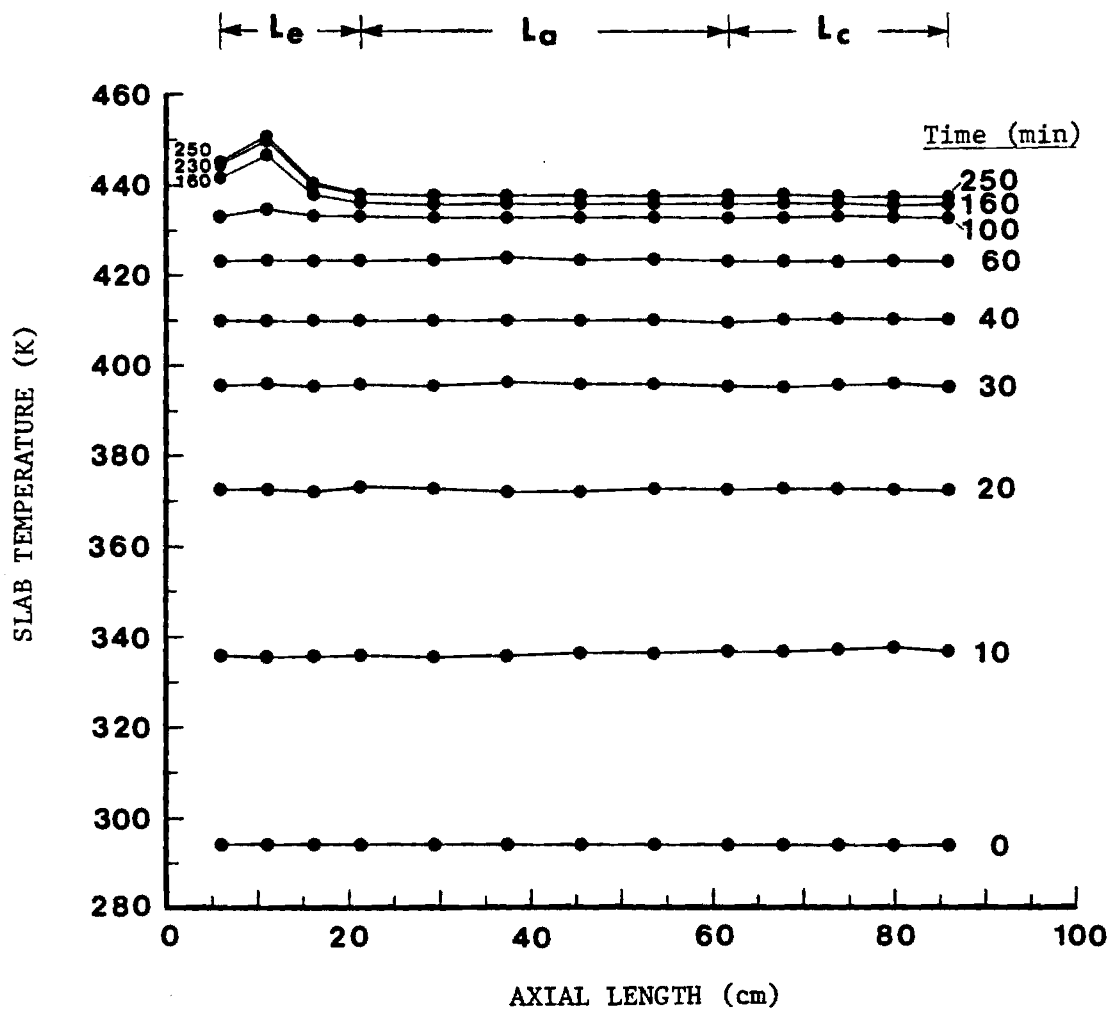
An example of the third type of transient behavior which may occur is shown in Figure 4. In this case the heat pipe was initially isothermal at room temperature while transporting no energy. At time zero heat is suddenly applied to the evaporator section while coolant flow is started through the cooling jacket. The device operates normally as a heat pipe for about fifty minutes being nearly isothermal at any

(Run 1-10)

Q_c = 131.6 W

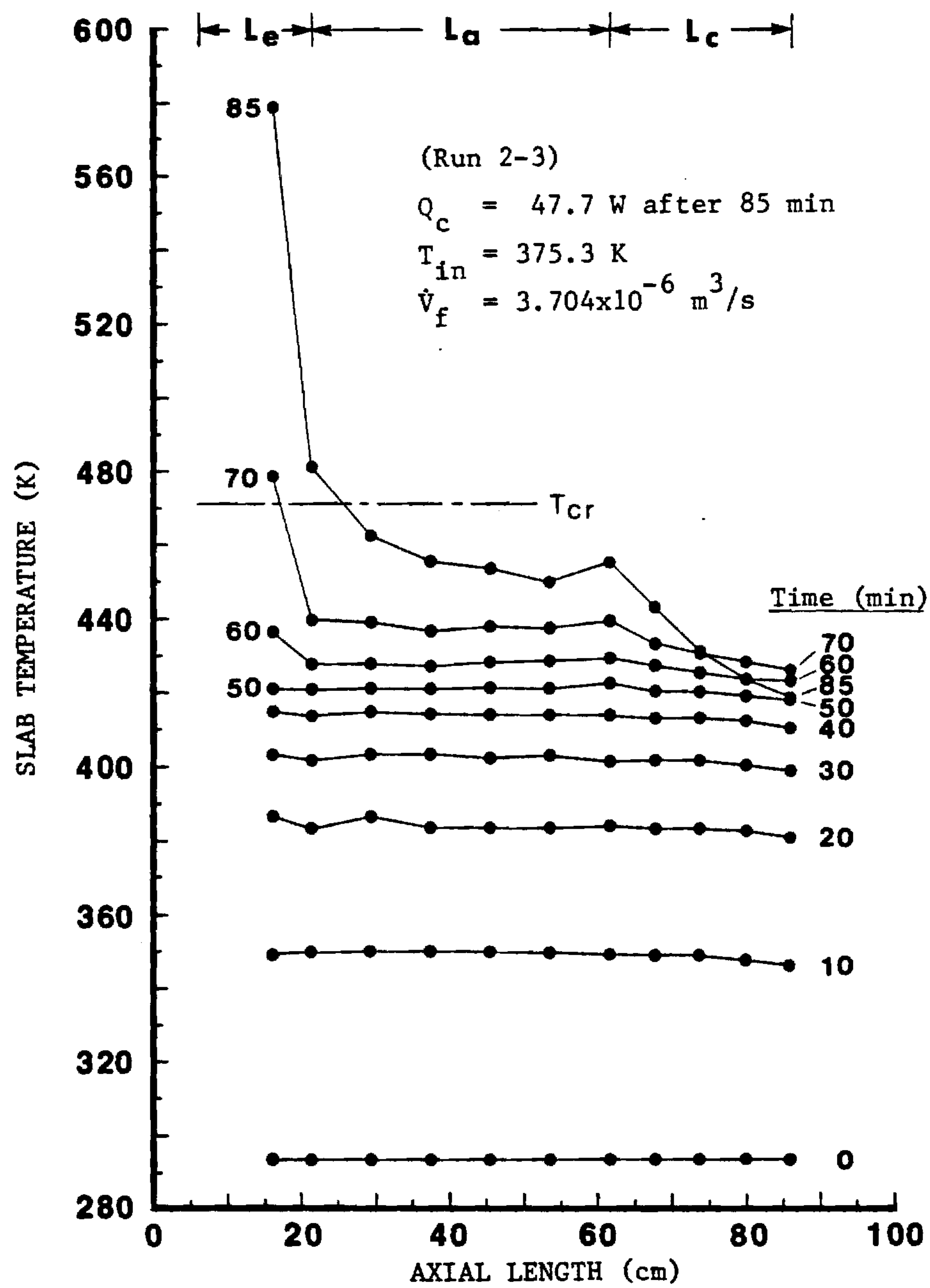
T_{in} = 346.1 K

V_f = 2.222×10^{-6} m^3/s



Slab Temperature Distribution (Run 1-10)

Figure 3



Slab Temperature Distribution (Run 2-3)

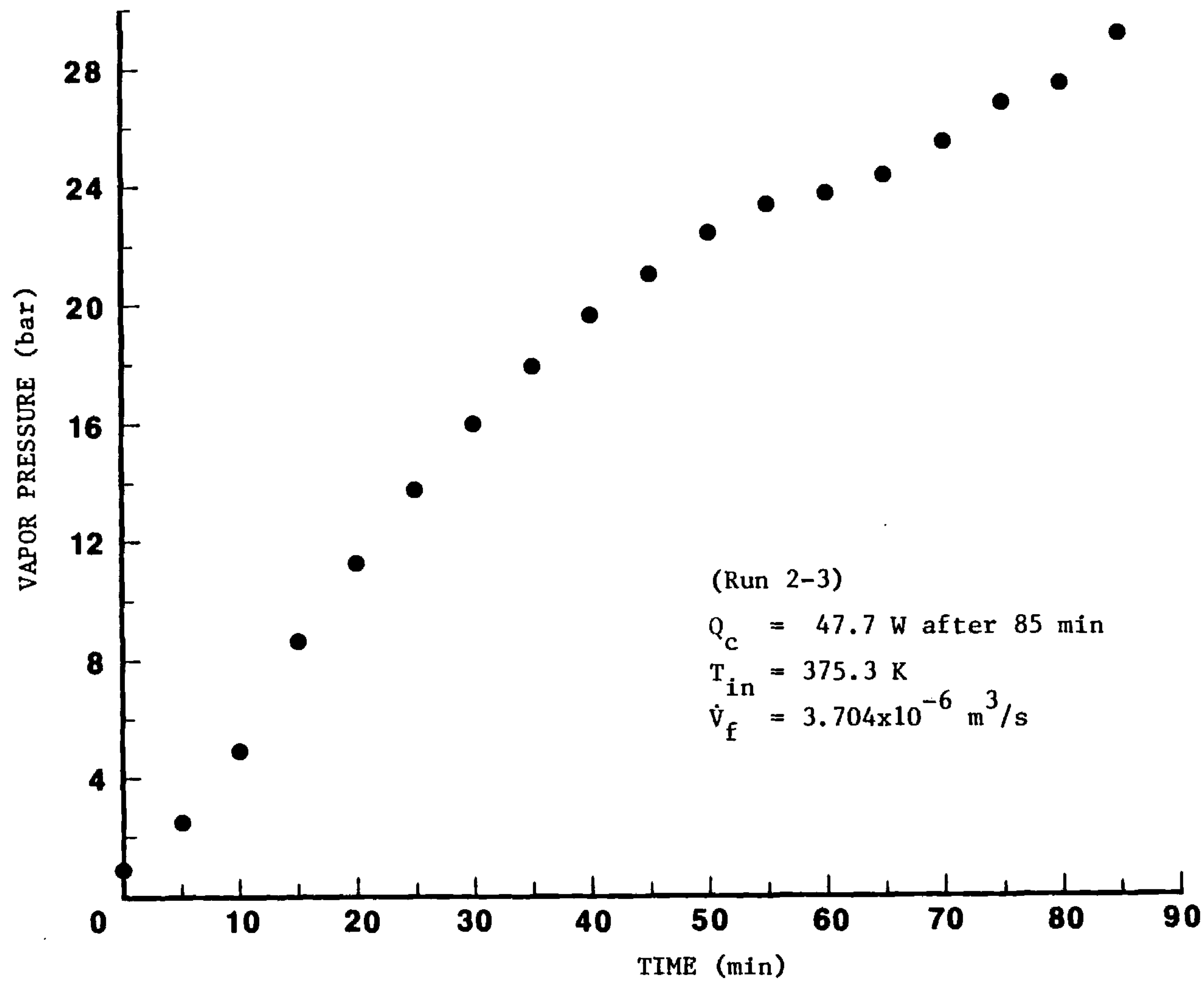
Figure 4

instant of time. Drying begins at about sixty minutes and continues until the heat pipe has failed at about eighty five minutes. Figure 5 shows that internal pressure has not stabilized and indeed is increasing very rapidly after eighty five minutes.

The fourth category of transients includes those cases where a heat pipe is restarted after initially experiencing partial or total drying of the capillary structure and subsequent failure. Figure 6 shows performance of the heat pipe during a restart after a failure similar to the one indicated in Figures 4 and 5. At time zero in Figure 6 input power and coolant flow were stopped after a failure for a period of fifteen minutes. During this period temperatures along the length of the Capillary structive began to approach a single value near the critical temperature. At a time of fifteen minutes, coolant was suddenly introduced to the cooling jacket. The heat pipe restarted as evidenced by decrease in axial temperature differences and the rapid decrease in internal pressure as shown in Figure 7. The restart shown in Figures 6 and 7 was from a condition where the evaporator and a portion of the adiabatic section were dried. In other cases where more of the capillary structure was initially dried, startup required much longer.

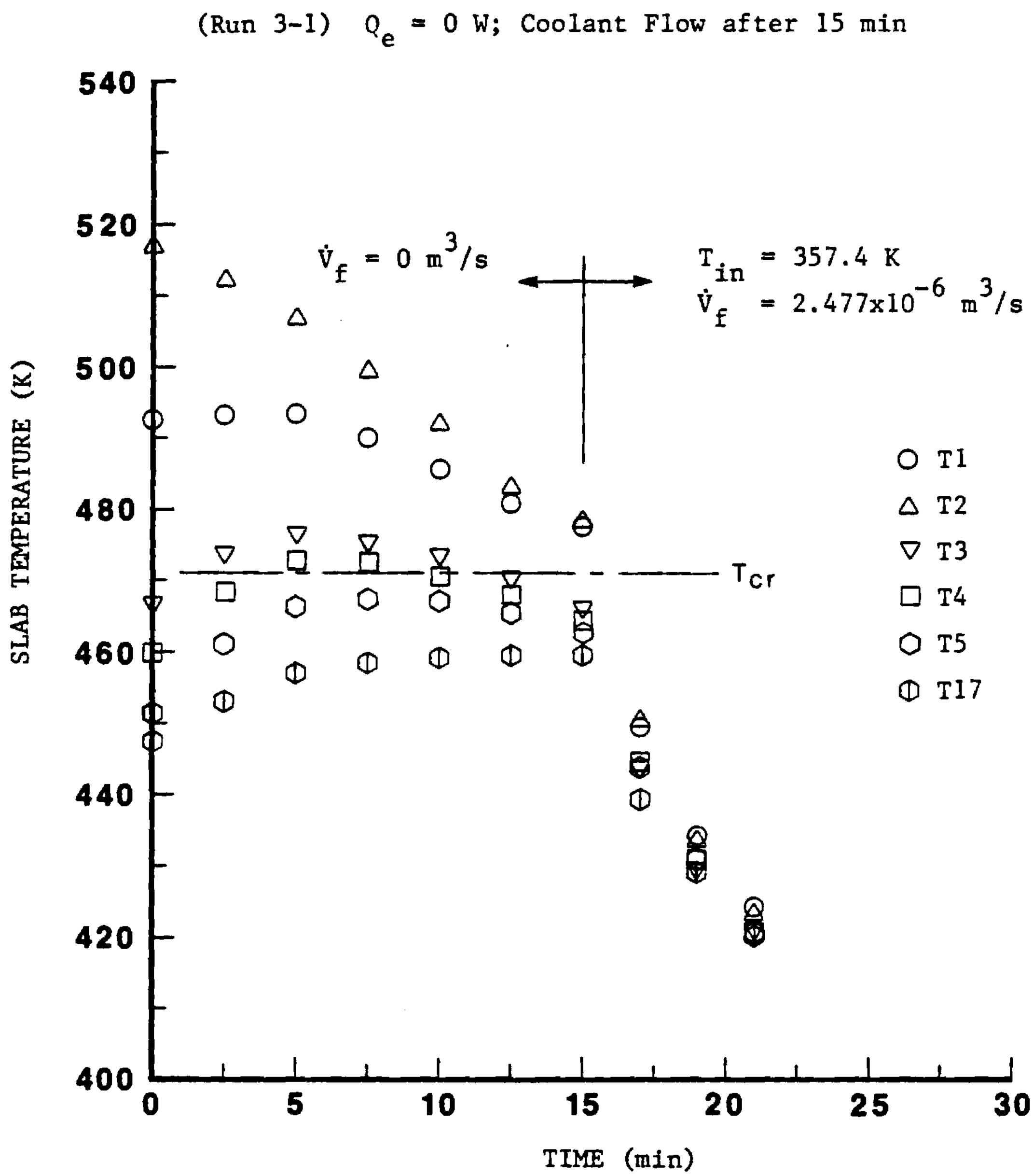
Computer Studies

A mathematical model and associated computer program intended for use in predicting transient operation of heat pipes has been under continued development in the heat pipe



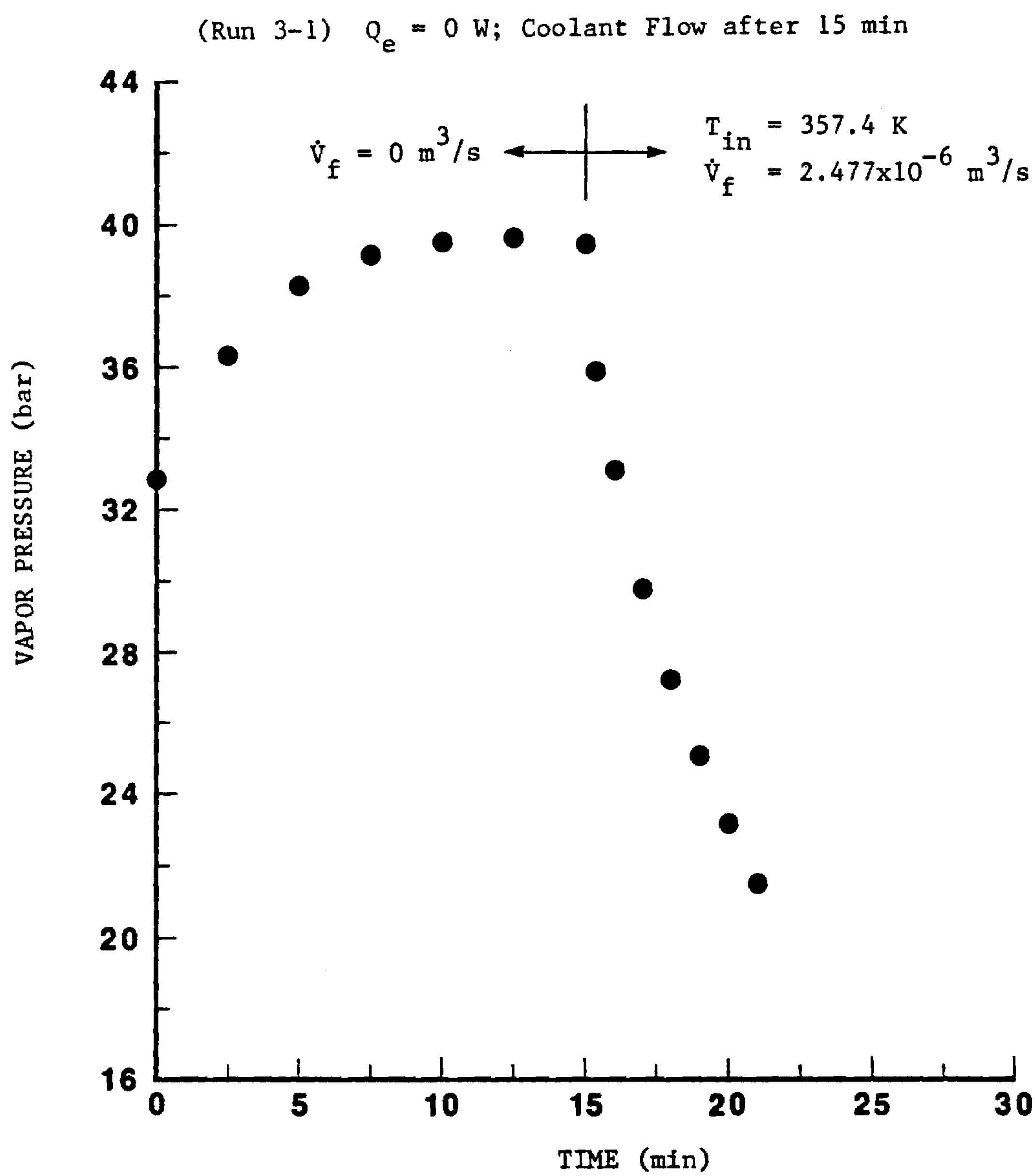
Transient Pressure Response (Run 2-3)

Figure 5



Transient Slab Temperature Response (Run 3-1)

Figure 6



Transient Pressure Response (Run 3-1)

Figure 7

laboratory at Georgia Tech for about ten years. A two dimensional nodal system, as shown in Figure 8, is used to represent the heat pipe. Radiation, conduction, or convection can be used to connect the heat pipe to the source and sink. Governing equations take account of solid and fluid property variations and liquid dynamics. Appropriate boundary and initial conditions and geometrical parameters are included as input and the governing equations written in finite difference form are solved using alternating implicit - explicit techniques.

Figure 9 shows a computed vapor temperature profile during a restart under conditions similar to those which prevailed for the case shown in Figure 6. The symbol β as shown in Figure 9 represents assumed clearances between layers of screen in the evaporator and condenser sections.

Correlation Equations

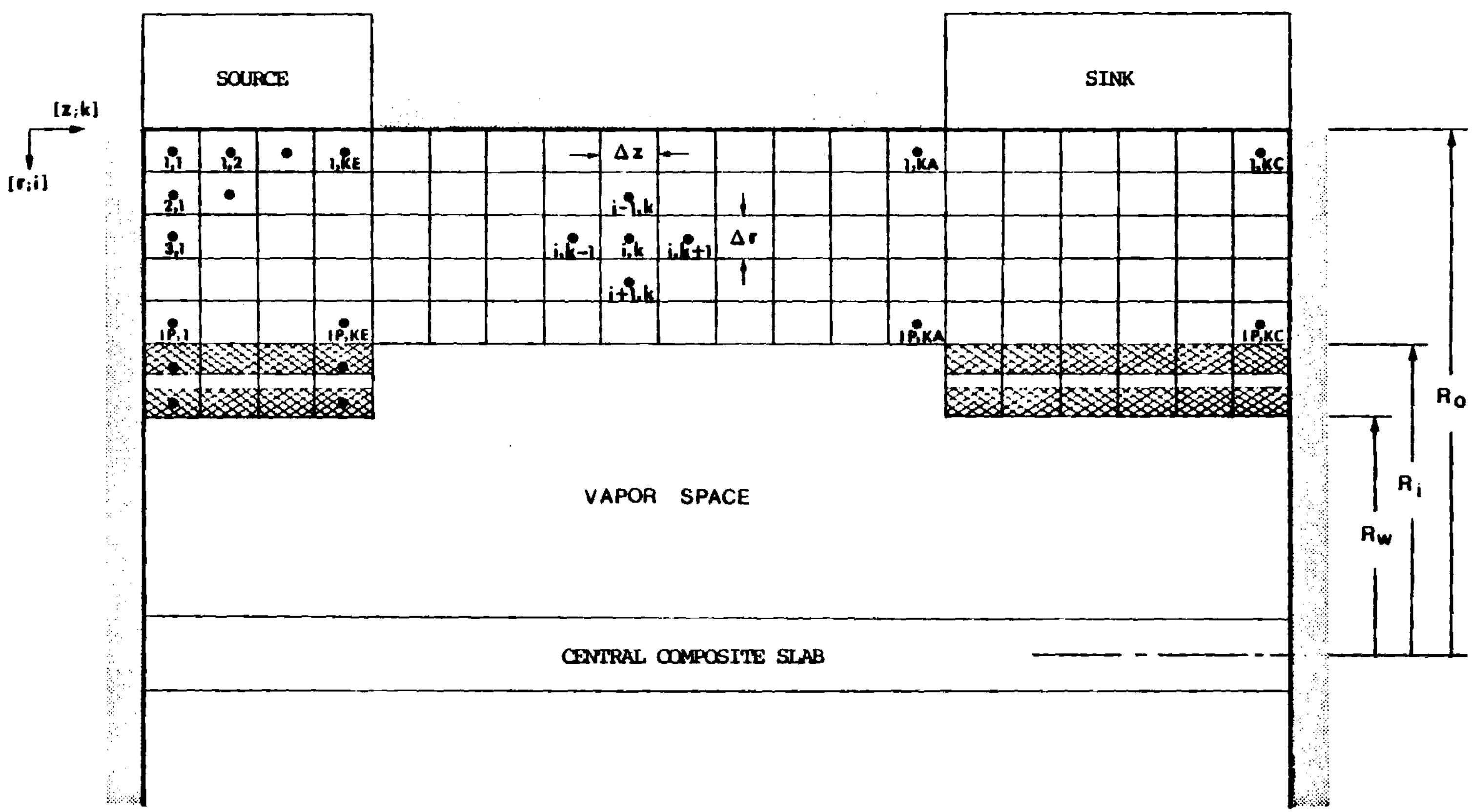
Under "normal" heat pipe transients where the capillary structure remains always fully wetted and the heat pipe is essentially internally isothermal at an instant of time, the performance can be approximated using a simple lumped mass model such as:

$$T_P = \frac{T_{PO}e^{-bt} + \left[\frac{R_P Q}{2} + ahAR_p \; T_i\right]\left[1 - e^{-bt}\right]}{1 - 2a\left[1 - e^{-bt}\right]}$$

where

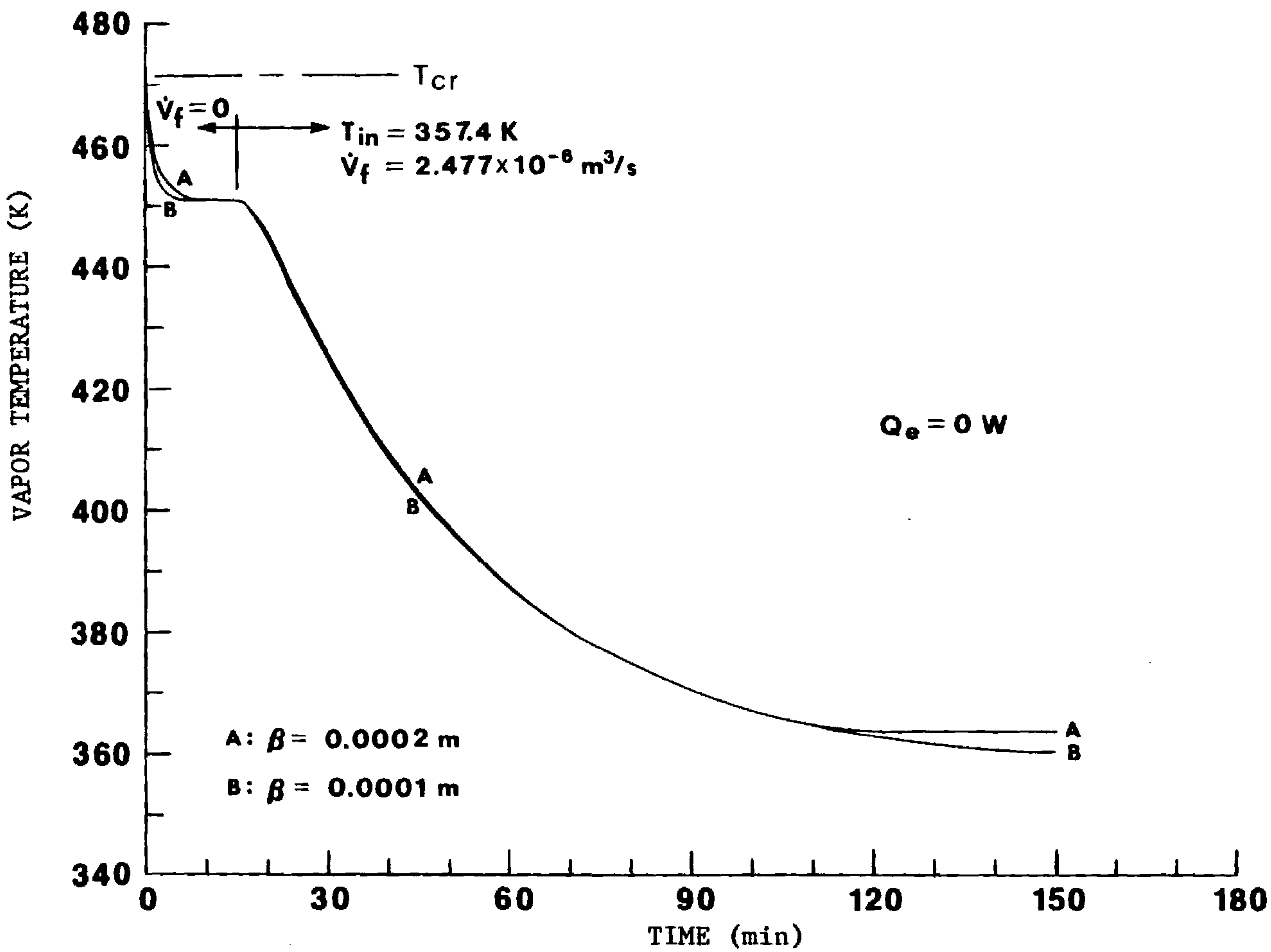
T_P = internal temperature

T_{PO} = initial internal temperature



Nodal System of the Heat Pipe

Figure 8



Transient Vapor Temperature Response from Theoretical Results (Run 3-1)

Figure 9

R_p = thermal resistance of heat pipe

Q = applied heat flux

hA = heat removal capacity of cooling jacket

T_i = temperature at inlet to cooling jacket

$$a = \frac{1}{hAR_p + 2}$$

$$b = \frac{2}{C_p R_p}$$

C_p = heat pipe thermal capacitance

The solid curves shown on Figure 2 were computed using this cimple relationship.

It is much more difficult to model, in simple form, category two transients where a portion of the evaporator capillary structure dries but where the heat pipe is still able to approach a steady operation. In these cases the heat pipe is not internally isothermal at an instant of time and the overall thermal resistance changes dramatically with time. Assuming that for a fixed heat input to the outside of the evaporator surface the heat conducted axially equals the heat input less the heat removed from the condenser, transient temperature distributions may be determined from

$$\theta_1(\phi,\tau_1) = \frac{L_e}{r}\left[\frac{\text{Coth } m}{m} + \frac{1}{2} - \frac{\phi^2}{2}\right.$$

$$- 2\sum_{n=1}^{\text{ODD}}\left\{\left(\frac{\text{Coth } m}{m} + \frac{1}{\lambda_n^2}\right)\frac{\sin \lambda_n}{\lambda_n} - \frac{\cos \lambda_n}{\lambda_n^2}\right.$$

$$\left.\left. + \frac{1}{\lambda_n^2}\right\}\cos(\lambda_n \phi)\, e^{-\lambda_n^2 \tau 1}\right]$$

and

$$\theta_2\ (\eta,\tau_2) = \frac{L_e}{r}\ \frac{\text{Cosh } m\ \eta}{m\ \text{Sinh } m}$$

$$- 2 \sum_{n=1}^{\text{ODD}} \left\{ \frac{\cos\ (w_n \eta)}{(m^2 + w_n^{\ 2}\ )} \left[\frac{w_n\ \text{Sin } w_n}{m\ \tanh\ m} \right.\right.$$

$$\left.\left. - (1 - \text{Cos } w_n) \right] \quad \exp\left[- (m^2 + w_n^{\ 2})\ \tau_2 \right] \right\}$$

where

θ_1 = temperature distribution in evaporator section

$\dfrac{T(x,t) - T_\infty)k}{\dot{q}_e be}$

θ_2 = temperature distribution in the adiabatic and condenser section

$\phi = \dfrac{X}{L_e}$, $\eta = \dfrac{X}{L}$

$\tau_1 = \dfrac{\alpha t}{L^2_e}$, $\tau_2 = \dfrac{\alpha t}{L^2}$

$\dot{q}_e$ = heat added less heat out of condenser

L_e = evaporator length

L = condenser plus adiabatic length

α = heat pipe thermal diffusivity

r = diameter of pipe

$m = \sqrt{\dfrac{hL^2}{kr}}$

h = film coefficient in condenser section

k = conductivity of heat pipe cross-section

$$\lambda_n = \left[w_n^{\,2} + m^2 \right]^{1/2}$$

and

$$\frac{w_n}{(w_n^{\,2} + m^2)^{1/2}} = - \frac{\tan \, (w_n^{\,2} + m^2)^{1/2}}{\tan \, w_n}$$

During a restarting operation after the capillary structure has completely dried or during startup from the supercritical state, liquid flow in the capillary structure controls the operation of the heat pipe. In most cases it is not possible to add large amounts of thermal energy at the evaportor end during the resetting process. The time for rewetting assuming constant heat addition q' and sudden cooling at time zero is:

$$t = \left(\frac{1 + N_{CO}}{N_q} \right) \left[\frac{L_c(L_o - L) + (L^2 - L_o^{\,2})}{L_q + LL_C - L^2} \right]$$

where

$$N_{CO} = \frac{A_p \rho_p C_p (T_{eo} - T_s)}{\varepsilon \; A_w \; \rho_\ell \; h_{fg}}$$

$$N_q = \frac{\pi \; do \; q'}{\varepsilon \; A_w \; \rho_\ell \; h_{fg}}$$

L_C = length of condenser

L_o = initial wetted length

L = wetted length

$$L_q = \frac{4\sigma}{r_p \mu \bar{K} \; N_q} + L_a(L_a + L_c)$$

L_a = length of adiabatic section

q' = heat input

T_{eo} = initial temperature in evaporator

T_s = saturation temperature

A_p = pipe wall crossectional area

ρ_ρ = pipe wall density

C_p = pipe wall heat capacity

ε = porosity

A_w = wick crossectional area

ρ_ℓ = liquid density

h_{fg} = heat of vaporization

do = outside diameter

r_p = screen pore radius

III. f. Significant Research Accomplishments

The broad goal of this project was to study experimentally and theoretically transient operation of heat pipes and to suggest some more or less general correlation relationships which might be used to predict performance. Very detailed internal measurements have been performed and a large quanity of basic data have been recorded for heat pipe operation under a variety of transient operating conditions. These data have been reduced in a number of ways so that general characteristics can be determined.

A mathematical model and associated computer program has been developed which is capable of predicting rather well the transient operation of heat pipes under many transient operating modes. This program has been used to perform parametric studies of the problem.

Based on a study of reduced experimental data and the results of parametric computer studies it has been found that at least four unique modes of transient operation may occur. In the first case we have "normal" transients wherein the capillary structure remains fully wetted at all times and internally the heat pipe is nearly isothermal at each instant of time. A simple correlation equation based on a single lumped thermal mass with variable properties predicts operation with relatively good accuracy. In the second category some drying of the capillary structure occurs in the evaporator zone but the heat pipe eventually approaches steady operation. Large internal temperature gradients may occur in this case and correlating results becomes much more difficult than in the first case described. Expressions have been developed which may be used to estimate transient internal temperature distributions.

The third type of operation includes those situations where the capillary structure in the evaporator zone eventually completely dries. When this occurs, the evaporator and condenser sections become almost thermally isolated from each other and internal temperatures and pressures grow rapidly to excessive values. One can predict when this will occur by simply comparing heat transport with capillary limitations.

The fourth class of operation covers startup after a capillary failure or startup from a supercritical state. Transient operation under these conditions is largely controlled by liquid dynamics in the capillary structure as it rewetts. Up to the time that liquid reaches the evaporator section, the evaporator and condenser sections are almost thermally isolated. Correlation equations for this class of problems are based on solutions of simple mass, momentum, and energy balances at the liquid-vapor interface in the capillary structure as liquid moves towards the evaporator.

It is believed that significant progress has been made during this project in understanding the transient operating characteristics of heat pipes. Also it is important to note that much of the data and some of the models developed will be of use in other fields where capillary action is important.

The results of these studies are being reported in the open scientific literature.

Several Ph.D. and M.S. candidates have and are using this project as a source for thesis work. They are receiving excellent training in the area of thermodynamics, heat transfer, fluid mechanics, material sciences and mathematics while performing research work within this program.

III. g. Personnel Supported

Dr. Gene T. Colwell has been supported approximately 25% time during one academic year and 66% time during one summer term. He has been responsible for overall direction of the project and has guided the day to day experimental and theoretical work.

Mr. John M. Hill has been supported approximately 33% time for one year. His primary responsbility has been to reduce theoretical and experimental data and to develop correlation expressions. It is expected that Mr. Hill will receive a Ph.D. degree in early 1982. His thesis is devoted to this project.

Other graduate students have worked on this project and written theses on the subject matter but have been supported from other funds. Dr. Won Son Chang received a Ph.D. degree in the Spring of 1981, Mr. Thomas Chitty received an M.S. degree in the Spring of 1981, and Mr. Ender Finol received an M.S. degree in the Summer of 1981

III. a. Abstracts of Theses

Won Soon Chang - Ph.D.

Thomas Chitty - M.S.

Ender Finol - M.S.

HEAT PIPE STARTUP FROM THE SUPERCRITICAL STATE

A THESIS

Presented to

The Faculty of the Division of Graduate Studies

By

Won Soon Chang

In Partial Fulfillment

of the Requirements for the Degree

Doctor of Philosophy

in the School of Mechanical Engineering

Georgia Institute of Technology

March, 1981

SUMMARY

The transient operation of a Freon-11 heat pipe at temperatures below and near the critical point has been studied theoretically and experimentally. A well instrumented test section has been designed and fabricated. The test heat pipe has been operated below and in the vicinity of the critical point, and detailed transient measurements have been made both internally and externally for a variety of operating conditions. In the heat pipe regimes the device operated very stably under transient and steady conditions. All inside temperatures in the capillary structure and the vapor space were almost uniform (within 1 K). When high heat load and coolant temperature were applied to the heat pipe, very large temperature gradients occurred throughout, internally and externally. Part of the evaporator was eventually dried out and with more thermal loading the whole capillary structure of the evaporator was dried. The internal temperature distribution, thereafter, diverged along the length of the pipe with increased evaporator temperature exceeding the critical and with descreased condenser temperature approaching the sink temperature.

Experiments on cooldown from the supercritical state have been carried out to determine the rewetting characteristics of the capillary structure. When part of the evaporator

is initially dried and temperature exceeds the critical value, the capillary structure will be rewetted and the heat pipe can be restarted normally. When the entire evaporator is initially dried and taken above the critical temperature, rewetting may occur but the device may not function properly due to remaining large temperature gradients. The heat pipe can thus be restarted successfully only when a small temperature gradient is established along the capillary structure.

A computational model has been developed based on the finite difference techniques for predicting the transient operating characteristics of low-temperature heat pipes. Variations of thermal properties have been taken into account by the use of cubic spline interpolation and an alternating-direction implicit method has been used to calculate the nodal temperatures. To simulate startup from the supercritical state, a simplified cooldown model has been developed using bulk fluid movement with phase-change at the liquid front. Good agreement has been obtained between the predicted and experimental values for normal operating conditions. There is a discrepancy between these values for cooldown from the supercritical state, mainly because the section between the evaporator and the condenser is assumed to be adiabatic in the model. Several different fluid gaps in the circumferential wick of the capillary structure have been incorporated in the model and the effects determined.

Unique transient experimental data has been obtained which will hopefully be of great use to designers of heat pipes and heat exchanger èquipment and to researchers in other scientific and engineering branches where transient capillary action is important. It is also hoped that the theoretical portion of this study will be of value in these fields.

HEAT PIPE PERFORMANCE
IN THE NEAR-CRITICAL REGIME

A THESIS

Presented to

The Faculty of the Division of Graduate Studies

by

Thomas C. Chitty, Jr.

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Mechanical Engineering

Georgia Institute of Technology

March, 1981

SUMMARY

This thesis consists of a study of heat pipe operation near the critical point. As a fluid approaches its critical point, several anomalies occur in the thermophysical properties. Several researchers have noted that the surface tension and heat of vaporization of a fluid go to zero at the critical point, while the film coefficient and constant pressure specific heat become very large. These phenomena cause heat pipes to exhibit strange behavior near the critical point, and because of this, predicting heat pipe operating characteristics near the critical point is difficult. This study is an attempt to determine if the unusual behavior of heat pipes near the critical point can be explained in the variations of the fluid's thermophysical properties.

Data taken with a refrigerant 11 heat pipe was used to make calculations of dimensionless parameters such as Reynolds and Nusselt numbers, as well as heat fluxes, pressure and temperature variations, and vapor velocities. Data was taken with the heat pipe as it was heated to the near-critical point, and as the heat pipe cooled down from above the critical point. Empirical equations and known correlations were then used to make calculations of velocities, interfacial resistances, vapor pressure drops,

heat fluxes, and other useful parameters. Comparisons of critical nucleate boiling heat fluxes and actual heat fluxes were also made.

Based upon analysis of the results, several conclusions were reached. As the temperature increases, heat transfer increases to a maximum, and then begins decreasing as the temperature nears the critical point. There are two major factors causing this decrease. The first is the decrease in capillary pumping capability, which is caused by the surface tension approaching a value of zero at the critical point. Decreasing capillary pumping action caused a decrease in mass flow rate. The second factor is the actual heat flux approaching the critical nucleate boiling heat flux, which will also decrease the mass flow rate. The decrease in mass flow rate as the critical point is approached, combined with the heat of vaporization becoming zero at the critical point results in a decrease in heat transfer, even though the film coeffient becomes quite large near the critical point.

It was also noted that the vapor pressure drop can be neglected, as is often done, but not near the critical point. As the critical point is approached, the vapor pressure drop value approaches the capillary pressure rise value. When this occurs, the capillary structure pumping capability is limited, the wick begins to dry out, mass flow rate is reduced, and the heat transfer capability is reduced.

HEAT TRANSFER MECHANISM IN THE EVAPORATOR END OF A HEAT PIPE

by

Ender Finol A.

Submitted to

Dr. Gene T. Colwell

For

Mechanical Engineering 8501
(Special Problem)

Winter Quarter 1981

March 19, 1981

INTRODUCTION

In recent years a considerable amount of effort has gone into the development of a highly effective heat transfer device called the "Heat Pipe". The heat pipe is best defined as a self-contained device capable of transferring large quantities of heat from a heat source to a heat sink through a relatively small temperature gradient. The phenomena evaporation, mass transport in the vapor phase, condensation, heat transfer by conduction (through solids), heat transfer by convection (through liquids), and surface tension pumping of a liquid in a capillary wick are used to transfer latent vaporization heat continuously from one region to another.

A typical configuration for the device is a closed cylindrical container with an annular porous wick (i.e., capillary structure) adjacent to the inside cylinder wall and an open space along the axis of the cylinder. (Fig. 1.1.). The wick structure is saturated with a liquid. During steady-state operation, heat is added to the evaporator end by conduction through the cylinder wall resulting in vaporization of the liquid in the wick and a flow of vapor towards the condenser end through the hollow central region. Heat is removed at the condenser end resulting in condensation of the vapor. The condensate returns to the evaporator end through the wick by capillary action to complete the cycle.

Heat pipes are usually constructed with a relatively large open space for the vapor flow which results in a very low pressure gradient in the vapor and thus, during normal operation, in very low axial temperature gradients. This feature together with the absence of moving parts and the ability to operate in the absence of gravity has resulted in much recent work on the development of the heat pipe.

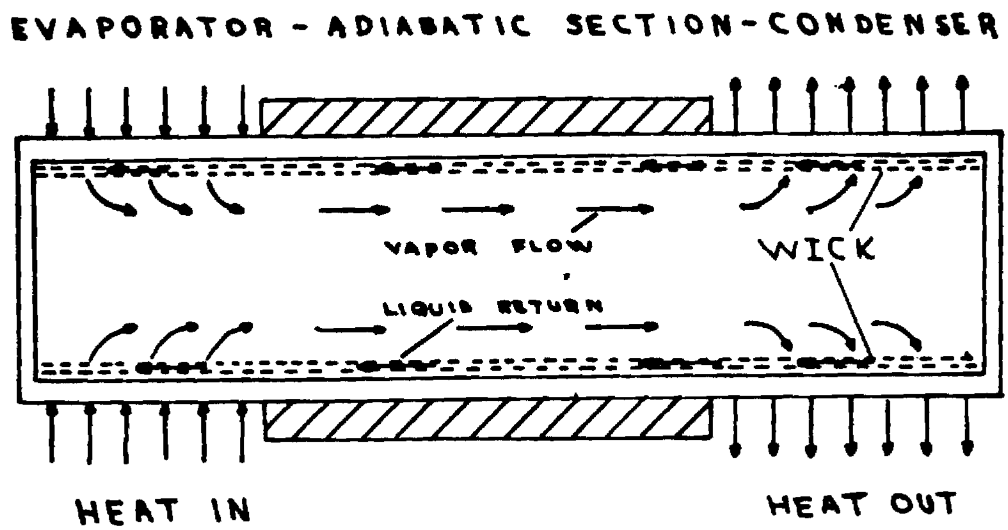


Figure 1.1. The Heat Pipe

While the heat pipe will operate as described above over a large range of conditions, eventually a heat transfer rate is reached where the wick in the evaporator section will dry out, resulting in a large axial temperature difference. (3 and 4)*. The most common reasons for the failure of a low temperature heat pipe are:

i. The wicking limit. The inability of the capillary forces acting in the wick to provide sufficient flow of liquid to the evaporator section. This results in a drying out of the wick adjacent to the heated surface, and an increase in the temperature of surface of the evaporator.

ii. A change in the mechanism of vaporization heat transfer in the wick resulting in the formation of a vapor blanket in the wick adjacent to the heated surface and having the same effect as (i). The boiling limit.

* Numbers in parentheses correspond to references listed at end of this work.

These limitations have been studied (analytically and experimentally) by several authors: Alleavitch (3), 1967; Johnson (4) 1971; Corman and Walmet (6), 1971; Alexander and Piver (16), 1972; Davis (5), 1974; Abhat and Seban (12), 1974; Tolubinskii et. al. (9, 10), 1979; Smirnov, (26), 1977; and B.A. Afanas'ev (27), 1979. However, the heat-exchange mechanism and the laws governing vapor formation in the wicks of low-temperature heat pipes have received no definitive explanation.

The objective of this study is to gain a better understanding of the heat transfer mechanism in the evaporator and investigating the influence of the convection on the heat transfer process in the wick structure at low and high heat flux densities.

EVALUATION OF THE THERMAL RESISTANCES IN THE HEAT PIPE

by

Ender Finol A.

Submitted to

Dr. Gene T. Colwell

for

Mechanical Engineering 8501
(Special Problem)

Georgia Institute of Technology
Spring Quarter 1981
June 1, 1981

1. Introduction

The objective of this paper is to evaluate the magnitude of the thermal resistances of each heat pipe component and its influence on the heat transfer rate. First, is discussed the pipe and wick geometry, the equations used for determining the thermodynamic properties of the materials utilized (Refrigerant-11, Silicone Fluid (SF1093), and 316 Stainless Steel), and the wire screen properties and geometry.

In the second section is presented the expressions for calculating the thermal resistance of each component of the heat pipe.

2. Heat Pipe Components

The following sections discuss the pipe geometry and general wick characteristics, and the variation of the thermodynamic properties of the materials used with the temperature.

2.1 Geometry and Dimensions of the Heat Pipe Considered in This Study

The heat pipe used in this study is shown in Figures 2.1, 2.2, and 2.3. This model was designed and fabricated by Chang [1]. The figures show the general configuration of the heat pipe and composite wick system.

The heat pipe consists of a stainless steel pipe of

HEAT TRANSFER IN THE CRITICAL REGION

by

Ender Finol A.

Submitted to

Dr. Gene T. Colwell

For

Mechanical Engineering 8501
(Special Problem)

Georgia Institute of Technology
Spring Quarter 1981
June 1, 1981

INTRODUCTION

Transport processes, particularly heat transfer, in the near critical region has been of interest for about the last fifteen years. There has been a steady development of steam plant towards supercritical conditions, and supercritical water has been considered as a coolant for several types of nuclear reactors. Helium is used at near critical conditions as a coolant for the conductors of electrical machines, and rocket motors are frequently cooled by pumping fuel (hydrogen) through cooling pipes at supercritical pressure.

Current emphasis stems from applications which require the use of a fluid in the near critical condition from inadequate information to produce satisfactory design expressions, and from an inadequate understanding of the mechanics which produce the peculiar behavior in the near-critical region.

The problem has been considered as one in which the variation of physical properties with temperature becomes extremely important. The effects and phenomena which appear in the processes near critical region have no counterpart with constant property fluids. At the same time experimental difficulties have hampered the investigation of these effects.

The near critical region may be though of as that region in which boiling and convection merge. Heat transfer near the critical point is taken to include boiling just below the critical pressure and convection just above.

This paper will be based on the region rather close to the critical point where the property variations are severe and where there are very significant heat transfer effects.

First, it is included a brief description of the behavior of thermodynamic and transport properties near the critical point. The equations of continuity, momentum, and energy are examined with a view to revealing the effect of variable properties. Criteria for the effects of bouyancy and acceleration are presented. Finally, this paper concludes with a review of the theories proposed to explain the possible heat transfer mechanisms under near critical conditions, together with suggestions for future experiments.

1. Physical Properties Near The Critical Point

The principal difference in the behavior of a fluid as its temperature is raised above the critical is evident from the pressure-volume diagram shown in Figure 1. Below the critical temperature the variation of pressure and volume along an isotherm shows discontinuities where the isotherm intersects the saturation line. Phase change occurs at the saturation line, and the horizontal constant pressure

III. b. Publication Citations

"Transient Heat Pipe Operation In The Near-Critical Region", Gene T. Colwell, Presented at IVth International Heat Pipe Conference, London, 1981 and Publication by Pergammon Press in "Advances in Heat Pipe Technology".

Two Journal papers are presently under preparation which cover the project.